DRIVE BASED VIBRATION REDUCTION FOR PRODUCTION MACHINES

A. Dietmair, A. Verl
Institute for Control Engineering of Machine Tools and Manufacturing Units (ISW), Universität Stuttgart, Germany

e-mail: anton.dietmair@isw.uni-stuttgart.de

Machine vibrations constitute a persistent challenge for high performance and high precision production machines that show resonant behaviour at some frequencies. Most remedies require additional actuators or precise knowledge of the dynamic behaviour of the machines under consideration. To overcome this limitation, we present several techniques that use the feed drives to dampen vibrations or avoid excitation and are applicable with minimal pre-knowledge. Moreover, we introduce an approach to treat machines with resonant frequencies that change with the tool centre point position.

Keywords
Machine Vibration, Active Damping, Control Algorithms

1. Motivation

Production machines traditionally achieve high productivity in conjunction with high precision through high mechanical stiffness and dynamic drives. An additional increase in the dynamic capabilities would require raising the motor power and stiffness even further. This leads to large moving masses and high power consumption and makes it impossible to answer the present demand for energy efficient machines. Although it is possible to feed back mechanical energy into the electric grid during braking, this always comes with energy losses. In addition to that, higher power motors often show higher power consumption for the same task than smaller motors.

The application of light weight design techniques to production machines opens up a way towards higher dynamics and reduced energy consumption at the same time [Zulaika 2006]. Namely finite element simulation and topological optimisation can contribute to minimising the moving mass while maintaining a mechanical behaviour that suits the application of the machine [Sekler 2007]. Taken to the extreme though, light weight structures often show a reduced stiffness and most importantly a very small mechanical damping. This leads to low dynamical accuracy, low process stability and low machined surface quality. Current research therefore aims to reduce vibrations in production machines [Dietmair 2007]. Often, additional damping modules [Ehmann 2004] or structurally integrated actuators [Zäh 2007] in autonomous control loops are proposed to achieve this goal. While active vibration absorbers can be tuned to a wide range of dynamic machine behaviour, semi-active systems require less energy for their operation [Kuhnen 2006]. Both lead to higher investment cost and an additional risk of failure.

Therefore, in the European Union framework programme 6 project EcoFIT, methods are developed that apply the feed drives of machine tools to reduce structural vibrations in production machines.

2. Drive Based Vibration Damping

Externally excited vibrations have to be damped by applying a counteracting force. While active damping modules are often collocated with the machine part to be damped, the feed drives act upon the machine structure from another position. Therefore, the characteristics of the transmission path have to be taken into account. Only vibration modes can be damped that are controllable by the feed drives and that are observable through the sensors available at the machine.

If these conditions are satisfied, the damping force can be calculated from the observed vibration and the transfer function between drive and structure. Basically any setpoint signal of the cascade control structure could then be modified by the damping signal, with only linear elements being required for the modification. As the actual vibration signal available on our experimental setup was a velocity signal and the control was best to be executed in the velocity control cycle time, we chose this loop of the cascade structure. The modification can then either be interpreted as an additive setpoint sequence, or as the computation of a synthetic velocity signal (Figure 1). Both interpretations are equal. Figure 2 gives an overview about the design procedure required for the application of the damping network to a specific machine.

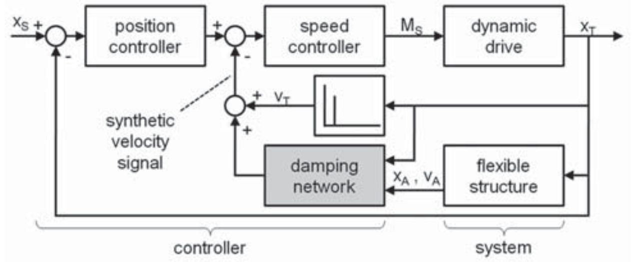

Figure 1. *The network for drive based damping in the velocity loop of the cascade control structure.*

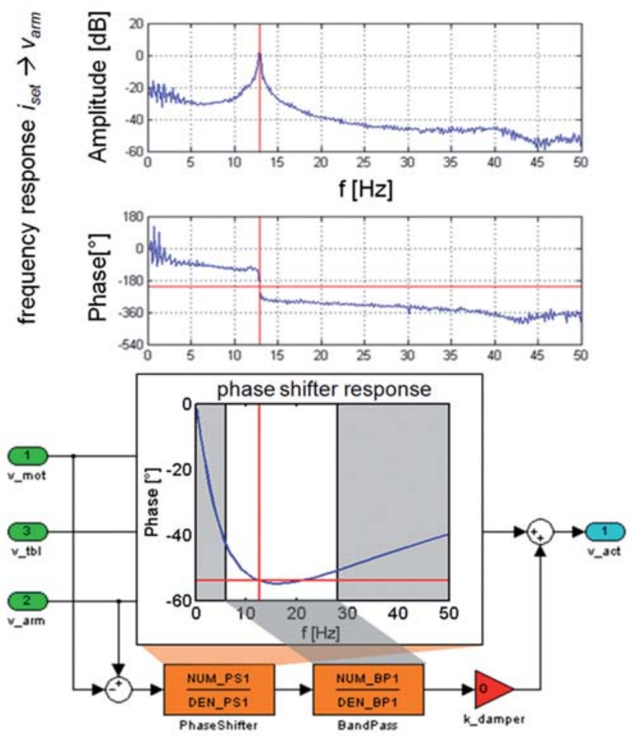

Figure 2. *Design of the phase shifting network for the calculation of the synthetic velocity feedback signal.*

First, the frequency response between force (or torque or current) and the velocity at the end point of the flexible structure to be damped is measured. This yields the resonant frequency and the phase shift at the resonance. A phase shifter, which may for example be an allpass, a digital filter, or a delay, is then tuned to give a total phase shift of 270° at the resonance. As phase shifting blocks can typically only be designed to match a desired profile in a certain frequency range, a band pass filter is then added to limit the actuation to an area around the resonance. As can be seen in Figure 3, even an im-

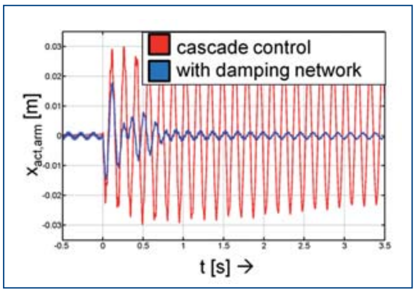

Figure 3. *Measured effect of the damping network.*

perfectly tuned damping network can greatly improve the vibration behaviour of a flexible structure.

In the example given there, the drive train of the testing bed under consideration showed considerable play. This led to a low amplitude limit cycle oscillation, as the phase shift is dependant on the amplitude. Therefore, we have to emphasise that nonlinearities limit the performance of the method proposed here.

Nevertheless, for compliant machines with a mostly linear behaviour, the benefits are clear. Flexible structures with low damping normally cannot be included into the position loop of a cascade controller, as they require a very low setting of the position control loop gain K_V (Figure 4, left). If drive based damping is applied, a much higher ***KV*** can be set, leading to a much better performance of the controlled system (Figure 4, right).

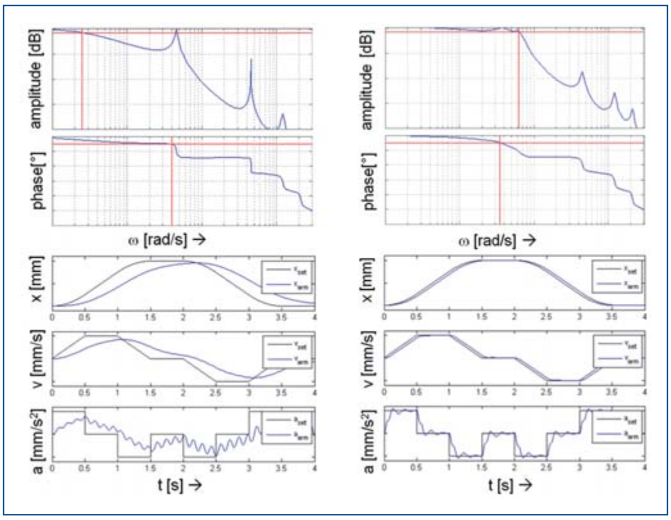

Figure 4. *Simulated behaviour of a directly position controlled flexible structure without (left) and with (right) drive based damping and maximum KV.*

While the plots shown her have been obtained from simulation, the results have been confirmed experimentally.

Active damping requires the vibration to be measured. Therefore, ongoing research is committed to finding affordable and reliable ways to measure or estimate the oscillation at the TCP, like for example an accelerometer based observer. Improved actuation concepts are also being studied, with inertial dampers and piezo actuators complementing the drive based damping in higher frequency ranges (Figure 5).

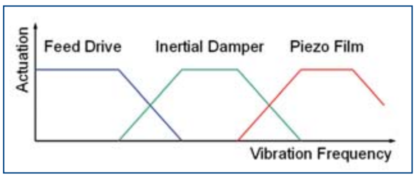

Figure 5. *Selective multi actuator vibration damping.*

3. Setpoint Filtering for Vibration Reduction

Sources for machine vibration excitation include the manufacturing process, eccentricities, nonlinear transmission elements, imperfections and collisions, all of which have to be damped actively as the often cannot be avoided entirely. A major source of vibration especially in fast moving high speed and high performance production machines can be manipulated almost arbitrarily, though: the path and velocity setpoint profiles machines are given to execute.

Consequentially, a number of setpoint filtering techniques have been proposed, some of which are available in industrial drive controllers. Most often, digital low pass or band reject filters are used [Economou 2000]. For compliant machines, low pass filtering leads to very low performance, though, and band reject filters can lead to overshoot. Other approaches include moving average filters [Jeon 2000], forming functions [Eloundou 2003] and input shaping with impulse sequences [Singer 1990]. Figure 6 shows the performance of input shaping and comb filtering to avoid vibration excitation of a beam structure. Setpoint filtering has the advantage, that they can be combined with existing setpoint interpolation software, but can lead to path deformation and limits to be violated.

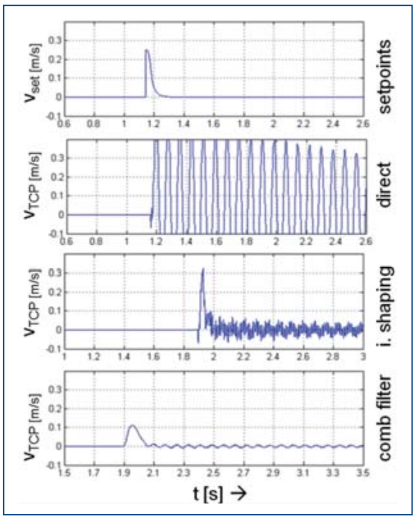

Figure 6. *Measured reaction of a flexible structure to a setpoint sequence (top) directly (2nd from top), with input shaping (3rd) and with comb filtering (bottom).*

4. Low Excitation Setpoint Profiles

Setpoint filtering seeks to remove unwanted frequencies from a given setpoint sequence. As a result, the path is deformed by the filters. This can be avoided, if the velocity profiles are generated analytically in a way that leads to gaps in the excitation spectrum that match the machine's resonant frequencies. For repetitive motions, this is an established technique that is referred to by the name of 'harmonic synthesis' [Dresig 2001]. Production machines are not always moved in a repetitive manner. In these cases, the principle of coun-

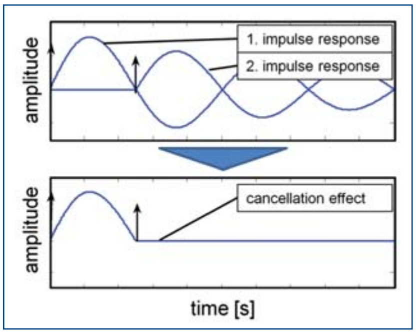

Figure 7. *Concept of counterpulse cancellation.*

terpulse cancellation can be applied to design excitation avoiding motion profiles (Figure 7).

Motion profiles for production machines typically consist of a sequence of discrete phases that can be chosen for their excitation to cancel with other phases. This applies, for example, to the jerk limited seven phase motion profile shown in Figure 8.

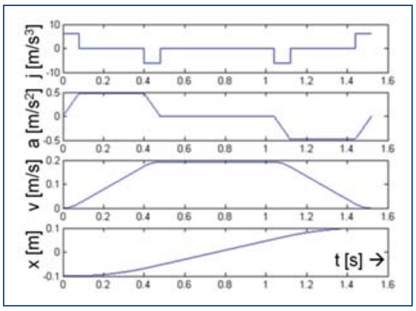

Figure 8. *Jerk limited motion profile.*

Based on the relative timing of the jerk steps, the excitation spectrum of such a profile always contains a number of gaps. Using an optimisation algorithm, the jerk steps can then be moved to shift these gaps to the resonant frequency of the production machine. Figure 9 shows the spectrum of a seven phase profile that has been optimised in a way that multiple gaps coincide at the machine's main resonance.

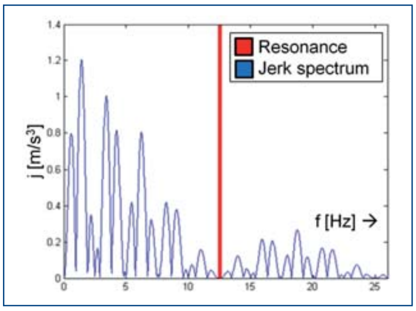

Figure 9. *Matching gaps in the motion profile jerk spectrum to the machine resonance.*

In the same way, the gaps could be shifted to accommodate multiple resonant frequencies. This hardly works in practice though, as the robustness is often too low. Moreover, it has to be highlighted, that the profile optimisation based on a Fourier spectrum is based on the assumption of zero damping. In cases where there is a small but significant damping, the counterpulses fail to compensate, as the previous pulse has already decayed to a certain extent. If the profile generator can be modified to allow for different settings of the jerk for each phase, this can be taken into account [Ladra 2007].

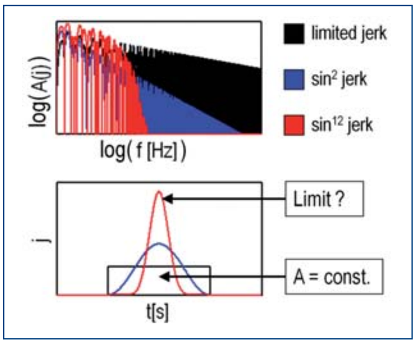

Figure 10. *Excitation spectrum (top) of jerk profiles of different degree of continuity (bottom).*

A complementary method to raise robustness and minimise excitation consists of the modification of the jerk profile itself to have a higher degree of continuity. Some numerical controls already provide sine-squared profiles, and other forms are possible (Figure 10). Care has to be taken about the maximum required jerk.

5. Effects of Nonlinearities

Up to now, all considerations have been based on a linear system formulation. Real production machines, though, do not fit this idealistic picture. Nonlinear transmission elements or nonlinear dynamic and static system characteristics, like for example play or friction, may distort the motions, thereby creating additional spectral content. In some cases, if these effects are known, they can be compensated in the control so that the controlled behaviour becomes linear [Kamalzadeh 2007].

Alternatively, mathematical optimisation techniques can be used to take into account invariant nonidealities and external disturbances. In these cases, the optimal motion profile for the linear case, e. g. that described in Section 7, is taken as a starting point for an optimisation algorithm that uses a simulation of the system behaviour to assess the quality of each modified profile.

Model predictive control can be seen as an extension to that which applies simulation based optimisation at each time step to compute the next setpoint [Altenburger 2004]. Unfortunately, the computational resources available in machine and drive control today are not sufficient for model predictive control.

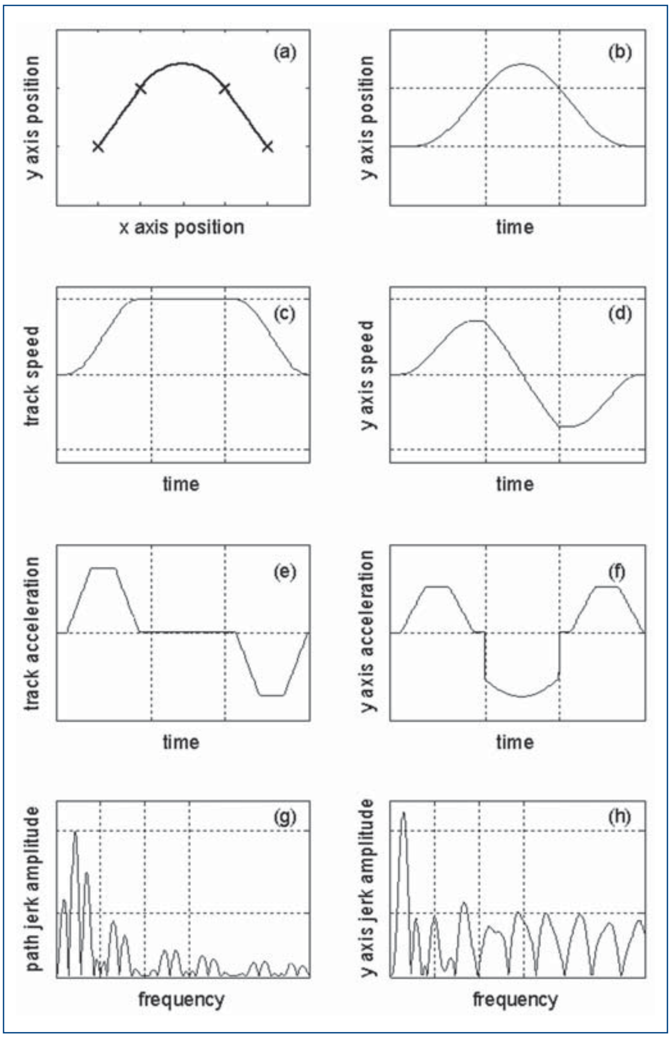

Figure 11. *Differences between track (left) and axis (right) motion for a simple continuous path execution.*

The path speed profile is transformed by the geometric path definition, mapping the track speed to parts parallel to the Cartesian space axes, and subsequently by the kinematic transformation of the machine into the individual axis speed profiles.

As can be seen from Figure 11, even for a perfectly linear machine and an optimised path speed profile (c, e), additional excitation (h) that is not present in the path motion spectrum (g) is caused by discontinuities in the axis motion (b, d, f) at numerical control program block boundaries (a). This effect becomes even more dramatic in five axis high speed freeform milling, where curved tracks are often approximated by a sequence of linear motions.

Even smooth motion trajectories can cause additional excitation due to distortion and modulation of the track motion spectrum. This becomes apparent by taking a look at the Fourier transform of the path speed x component as a function of the track position parameter s and the track velocity $v(t)$:

$$\begin{aligned} F_v(\omega) &= \int_{-\infty}^{\infty} \left.\frac{dx}{ds}\right|_{\bar{x}(t)} v(t)\left(\cos\omega t - j\sin\omega t\right)dt \\ &\neq \left.\frac{dx}{ds}\right|_{\bar{x}(t)} \cdot \int_{-\infty}^{\infty} v(t)\left(\cos\omega t - j\sin\omega t\right)dt \end{aligned} \tag{1}$$

It is not possible to separate path and track speed in general. Instead, each type of path has to be considered individually for its effect on the spectrum. For example, if the tool is moving on a circular path, the side bands are well known to be spaced symmetrically around the original track speed frequencies with a distance corresponding to the current circle frequency.

For polynomial formulations like spline space profiles, it is possible to break equation 1 down into multiple additive parts and the side bands can be computed analytically. To do this, an explicit parametric form of the spline path polynomials has to be available. A parabolic example is given in equation 2:

$$\begin{aligned} F_v(\omega) &= \int_{-\infty}^{\infty} \frac{d}{ds}\left.\left(c_1 s^2 + c_2 s + c_3\right)\right|_{s(t)} v(t)(\ldots)dt \\ &= 2c_1 \cdot \int_{-\infty}^{\infty} s(t)\cdot v(t)\cdot\left(\cos\omega t - j\sin\omega t\right)dt \quad + \\ &\quad + \ c_2 \cdot \int_{-\infty}^{\infty} v(t)\cdot\left(\cos\omega t - j\sin\omega t\right)dt \end{aligned} \tag{2}$$

Here, the first term constitutes the modulation and the second term is the amplified original spectrum. From this formulation, we draw the conclusion that the problem of excitation avoidance can be broken down to tool path definition and subsequent velocity profile design to avoid indirect vibration excitation.

As an alternative, for nonlinear systems, flatness based control approaches have been studied [Verl 2008]. Based on this principle, a trajectory for all states at the same time and thus to define a motion that is free of oscillation can be specified. It is even possible to create a linearising controller cascade. Unfortunately, this technique requires a deep theoretical understanding, requires a detailed model of the system and is computationally intensive.

6. Parameter Scheduling

All vibration reduction techniques presented here are specifically tuned to the resonant frequencies of the machine to be controlled. This might be problematic, as real production machine often show a dynamic behaviour which is varying with time, e. g. due to temperature variations, or position of the axes. If these variations are small, the control algorithms can be tuned for robustness, but for light weight machines, the variations are expected to be substantial.

In these cases, it is necessary to adapt the parameters of the vibration control algorithms to the changing conditions. This could, in principle, be done with any of the large number of adaptive control techniques that can be found in the literature.

But all these techniques share the disadvantage of a time lag between changes in the system behaviour and the re-tuning of the controller. In addition to that, security requires changes to machine control algorithms to be deterministic, which cannot easily be fulfilled by adaptive techniques.

In the EcoFIT project, we therefore examine an alternative feedforward approach to parameter scheduling (Figure 12).

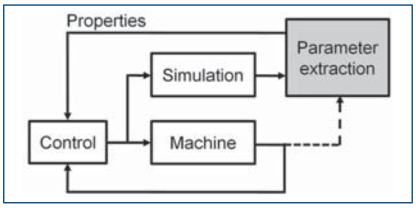

Figure 12. *Controller scheduling based on online parameter extraction from a simulation model.*

In its most basic form, measured characterristic maps could be used for the scheduling (dotted line) [Symens 2004], but these would require the machine to be measured at an enormous amount of positions. Instead, we seek to extract the system resonances from a simulation model in real time. Simulation models become increasingly available from the design stages and can be simplified to be computed in real-time [Sekler 2008]. These resonances are then used as an input to vibration damping and avoidance techniques.

Care has to be taken with respect to the may control parameters are scheduled. In some cases, moving between stable parameter settings can lead to instability of the system [Shamma 1992]. But preliminary studies on scheduling of standard cascade control parameters have shown that for continuous changes at rates lower than the oscillation speed, stability can be ensured (Figure 13).

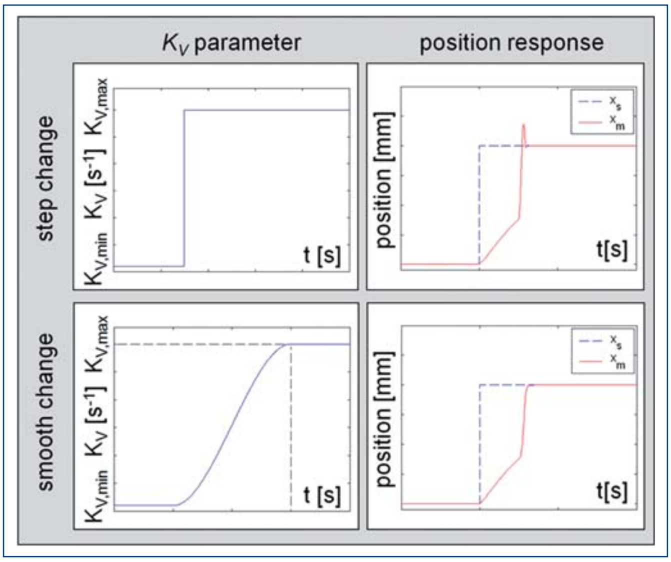

Figure 13. *Feed forward KV parameter scheduling.*

This condition can be maintained in practice, as the changes in machine properties occur only over large ranges of motion: In addition to that, the scheduling in our case is performed in a feed forward manner, and no potentially unstable adaptation feedback is present.

7. Conclusions

In this paper, we have presented practically applicable methods to tackle vibration problems in production machines using the feed drives as actuators. Active damping of external vibrations and avoidance of excitation through motion profile design for Cartesian machines with linear properties and motions have been at the heart of the presentation. An outlook has been given on the effect of nonlinearities in machine and path as well as variations of machine parameters. Future work will be directed towards these areas.

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Contacts:

Anton Dietmair, Universität Stuttgart
Institute for Control Engineering of Machine Tools
and Manufacturing Units (ISW), Germany
Seidenstraße 36, 701 74 Stuttgart
tel. +49 711 685 824 10, fax. +49 711 685 828 08
e-mail: anton. dietmair@isw. uni-stuttgart. de